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An Experimental Procedure to Test Sputtered Gold as a Solid Film Gear Tooth Lubricant.

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THESIS

AN EXPERIMENTAL PROCEDURE TO TEST SPUTTERED GOLD AS A SOLID FILM GEAR TOOTH LUBRICANT

by

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September 1968

[redacted]

AN EXPERIMENTAL PROCEDURE TO TEST SPUTTERED

GOLD AS A SOLID FILM GEAR TOOTH LUBRICANT

by

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Lieutenant, United States Navy
B.S.M.E., University of New Mexico, 1963

[redacted]

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ABSTRACT

The design, construction, and instrumentation of a solid film lubricant tester are discussed. The testing phase and evaluation of test data are mentioned. The device was constructed to test sputtered gold as a gear tooth lubricant for low vacuum operation. The unit is small and self-sustained for operation entirely within the vacuum chamber with only external electrical leads. Methods of measuring gear tooth frictional forces are discussed. A computer program is used to check the results of classical gear tooth strength theory.

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ACKNOWLEDGEMENTS

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1. Introduction.

The phenomenon of lubrication by a viscous medium between sliding and/or rolling contact machine elements is farily well understood today. However, the advent of space exploration has increased interest in the field of dry or solid film lubrication. The solid film lubrication field has no real theoretical back ground and has gradually evolved from many experimental studies. Products available today are those which performed well against others in the laboratories.

Many different types of solid film lubricants and surface bonding techniques have been tried. A quick review of recent literature will reveal the experimental results of a few of these, such as graphite, sulfides, carbides, and oxides, together with various binders and coating techniques; pure element coatings, plated elements, and so on. But, as far as the author knows, no experimental work has been done to test sputtered gold as a solid film lubricant.

The purpose of this work was to develop the necessary equipment to evaluate gold sputtered surfaces under various loadings with sliding and/or rolling contact. The test equipment is designed to give good reliable data from which a statistical analysis may be made to determine the useful life to failure of the gold film. Logically, the work may be broken into five categories as follows: 1) design, 2) construction, 3) instrumentation, 4) testing, and 5) evaluation. This paper will deal with only the first three major categories.

2. General Discussion.

If the normal method of lubrication by a viscous medium is required, a system operating at extremes of temperature, in hard vacuum, and in zero gravity fields must, by necessity, become extremely heavy and complicated. Problems such as sealing, artificial gravity fields, forced lubrication systems, element and lubricant temperature controls, etc., must be accounted for in the design. With this in mind, it is easy to see why the space systems to date have mostly been solid film lubricated. Since the lubricant is bonded directly to the contact surfaces, no supporting systems are needed. The resulting equipment may be smaller, lighter in weight, and less complex than a similar viscously lubricated system.

Immediately one may think of advantages in using solid film lubrication for other than space applications. Certainly there are many uses and solid film lubrication has almost limitless applications. But this paper will deal only with the lubrication problems found in the space industry with environments and pressures from atmospheric to 10^{-8} torr and temperatures from -250°F to +300°F.

The basic mechanism of solid film lubrication is direct contact between mating machine elements. Since the contacting surface or surfaces are covered with a film of some type, the harder subsurface layers of the machine elements are protected. However, because of this direct contact, some form of wearing away or shifting of the lubricating film will take place. Eventually this film protection will "fail" and expose the underlying surface machine elements to direct contact. Failure of the exposed parts soon is followed by contact welding or a similar failure mechanism.

Immediately it is apparent that solid film lubricants must possess two important properties. They must adhere to the contact surface very

tenaciously, thereby lowering the possibility of movement and wearing away. And they must have a reasonably low coefficient of friction.

Thus far, only advantages of solid film lubrication have been discussed. Certainly there are disadvantages. Because there is no circulating lubricant for cooling, the heat generated in the contact area may become a governing design factor. Generally, the coefficient of friction for solid film lubrication is higher than that for viscous lubrication. It is inevitable that because of direct contact the solid film used for lubrication will eventually wear away and fail. The upper limit on sliding velocities must be somewhat lower for solid film lubrication than for viscous lubrication.

Molybdenum disulfide in its various forms is probably the most popular dry film lubricant for space applications. However, both Young [13] and DiSapio [2], describe some of the deteriorating effects of conventional lubricants and other fluids on resin-bonded films. Elevated temperatures in the range of 750°F [5] also reduce the lubricating qualities of molyybdenum disulfide films. Flage [4] conducted experiments on a supposedly high temperature ceramic bonded film and found it unsatisfactory. These together with many other disadvantages, seriously limit the designer in his work with this lubricant. The design of lubrication systems for a ten year communication satellite or long term space probe are extremely difficult until new break throughs are accomplished in this field.

Sputtered gold promises to alleviate many of the problems encountered in the solid film lubricant field. The sputtering process is, itself, quite old and described by Strong [10]. Sputtered coatings have not been widely used until recently because of the excessive time involved in depositing the film. However, Varian [9] generally describes a recent industrial mass production process utilizing new equipment for this old

process. This reference shows that fast, rapid, sputtering on an economically competitive basis may be accomplished.

Many advantages could be realized with sputtered gold films. Some experimenters have compared coefficients of friction obtained with solid film lubricants against the relatively low coefficient of friction of gold. The requirements for the sputtering process are not very stringent and call for only a rough vacuum, about 2 microns, and only a moderately clean surface. The sputtered coating adheres exceptionally well to most metalic surfaces and is reported to be extremely wear resistant. Ordinary lubricants could be used in conjunction with the gold film without the deterious effects described with resin bonded films [13]. Low temperature operation would be almost unlimited; and high temperature operation limited only to the softening of the gold film or basic machine elements. Gold begins to soften at temperatures above 1450°F. Outgassing and evaporation of the sputtered gold film would not present any great problems. In conclusion, the sputtered gold could provide a good all purpose dry film lubricant without many of the disadvantages found in the solid film lubricants presently available.

Also worthy of mention is the method of analysis used for tooth strain gage positioning in Section 5 - Instrumentation. Classical methods were used to determine optimum strain gage location. Previous work by J. P. Malone [7] provided a computer program, utilizing the finite element method, which was then used as a check on the results obtained by classical methods. This program also provided additional information which proved helpful in optimizing the strain gage location. Some interesting results, comparing the classical gear tooth analysis, to the results obtained by computer, are found in Section 5.

3. Design.

The figures found on pages 29 through 35 are photo reproductions of some of the actual prints used to construct the device. The assembly drawing is located on page 27 . These drawings were necessarily reduced in size, some more than others, for inclusion in this paper. Therefore, the scales as noted should not be used directly. A picture of the constructed device is found on page 28.

During the preliminary design phase, it was decided to construct a device which would be both simple and versatile. Many gear and lubricant testing devices have been designed and built. They are usually so complicated and contain so many built-in variables that correlation of data from any two is extremely difficult. Figure 10 shows the basic principle of operation of the device under consideration. The gear teeth are loaded against the springs, which because of their size and spring constants, maintain relatively constant opposing force during small rotations. The spring forces are designated as F_s in Figure 10. The springs in turn transfer the torque to the shaft by means of the spring retainers shown as part number 4 on Figure 1. The shaft ends are fixed to the test device base. Only the shaded portion of the test gear shaft as shown in Figure 10 is in torsion. One motor is completely adjustable through 360 degrees to insure that the loadings in the testing sections can be equalized. Small oscillatory motions are desirable to test small portions of the solid film on each tooth. Time is not wasted by completely rotating the gears. During two motor operation only couples affect the test gear. Gear tooth separating forces balance each other and all other undesirable forces have been virtually eliminated. The test gear then "floats" on the shaft center.

As was previously mentioned, the pressure and temperature range of

the device would be ambient to 10^{-8} torr and -250°F to +300°F, respectively. The lower temperature limit was not attained as explained in Section 6 - Testing. Size was a controlling factor because the finished product had to fit in the Low Vacuum Cryo Chamber at the Naval Postgraduate School. The constructed unit measured 15" x 10½" x 6", well within the chamber limits. Weight of the device was not considered critical. The equipment will operate at any orientation. Throughout the design phase, device variables were limited only to essentials in order to obtain the most simple configuration possible which would accomplish the job.

Each motor may be operated independently or in parallel. This allows single or dual tests, at the same contact stress. With both motors operating in parallel, two different solid film lubricants may be tested and compared under identical conditions.

Stainless steel was utilized whereever possible in the design because of its noncorrosive, low temperature, and good outgassing properties. By utilizing the same material where ever possible, stray effects due to temperature are also minimized. It was found in all cases, except the springs, shaft, and gear teeth, that deflection governed the design rather than strength. Deflections in all cases were limited to the order of magnitude of the gear tooth errors, 0.0002 inches.

The gear or testing section was designed to give Hertzian contact stresses up to 200,000 psi. A diametral pitch of 12 was selected for the gear teeth in order to provide small teeth similar to those used in the space industry, yet large enough to instrument with standard strain gages. The teeth are full depth involute profiles with a 20 degree pressure angle. The material for the gears was AISI 4340 modified (vacuum melt), vice the 8640 specified on the drawings.

The gears have a surface hardness of Rockwell "C" = 14. Faires [3] lists an ultimate tensile strength of 122,000 psi for AISI 4340 cold drawn, one inch specimen and a hardness of Rockwell "C" = 33. The vacuum melt obviously differs from the cold drawn specimen and an ultimate tensile strength of 122,000 psi cannot be guaranteed for the gear and pinion. The gear and pinion drawings are found on pages 34 and 35.

Contact stresses as high as 500,000 psi Hertzian could be achieved by changing the size of the stepper motor pinions and torsion springs. This stress level could be met with the existing motors and equipment, except as indicated, without exceeding the 100% capacity of the device.

The stepper motors obtained for this project were manufactured by Hughes Aircraft Company. Similar motors were used in the Surveyor moon program, and were especially designed for vacuum operation. A more complete description of the motors is given in Section 4.

The helical torsion springs were designed according to the procedures outlined by Wahl [12]. The spring is shown in Figure 11. At a Hertzian contact stress of 200,000 psi on the gear tooth, the springs carry a moment of 372 in-lbs., and a flexural stress of 137,000 psi. It is easily seen that, except for the gear teeth in the test section, the springs are the most highly stressed and critical elements of the design.

The test gear shaft was designed using the strength and deflection formulas as found in Faires [3]. It was assumed that the shaft ends were built in connections. For two motor operation only torsion appears in the shaft between the spring retainers and shaft pillow blocks. For single motor operation this torsion is present, plus forces acting upon the shaft center. These forces produce bending of the shaft. Thus if one wishes to eliminate deflection of the shaft and test gear, two motor operation is preferable.

4. Construction.

The driving force for the lubricant tester is provided by two stepper motors. These motors were developed and manufactured by the Hughes Aircraft Company, Space Systems Division, El Segundo, California. Electromechanical Technology [6] gives a brief description of the motors together with some operational characteristics.

The bi-directional motors were designed primarily for use in the NASA Surveyor program but can be used, without modification, in the lubricant tester. However, special mountings are necessary because of the original installation method. The size of the motors is not critical for this application because they are extremely small and light weight for the output torque developed. Torques up to 400 inch-pounds may be obtained.

A floating solenoid is the primary driving force in the motor. Reciprocating linear motion is converted to rotary motion by a detent cam arrangement. The rotating output is then passed through a 289/1, triple reduction, plantary gear assembly. This produces 0.1423 degrees of rotation of the output shaft per input pulse or step. Power required is about 85 watts at the pulse peak for a 50 millisecond pulse. At ambient temperatures, pulse or step rates as high as 11 Hertz may be used for continuous operation [6].

Throughout the construction, these motors are made of low vapor pressure materials. Bearings and gear surfaces are coated with a special bonded dry film lubricant. The entire motor/gearhead is sealed to minimize the loss of trapped atmosphere during vacuum operation. Operation at pressures below 10^{-7} torr is possible. [6].

The springs and gears for the device were obtained through commercial sources. The springs were hot formed in accordance with figure 11. The

gears were manufactured in accordance with Figures 8 and 9. Maximum allowable tooth error and surface finish are equivalent to those found in precision gearing.

The remainder of the device was constructed in the machine shop of the Naval Postgraduate School. All blind stud holes were drilled to the outside of the member to prevent air entrainment in the vacuum chamber. After manufacture, all elements were cleaned with toluene and acetone to remove grease and other substances which might produce outgassing problems in the vacuum chamber.

Some gold sputtering was accomplished with equipment furnished by the Physics Department of the Naval Postgraduate School. An equipment set-up similar to that described by Belser and Hicklin [1] was utilized. A mechanical, rotary type, vacuum pump was more than sufficient to maintain the pressure in the bell jar at about 20 microns. A 3000 volt power supply was utilized in the sputtering process with currents in the range of 30 milliamps. A gold foil target of approximately 2 x 4 inches was placed about ¼ inch from the object to be sputtered.

This method of sputtering, called the "glo discharge method," produces the gold film at a rather slow rate. It is easy to see why newer and faster methods of sputtering, such as the "rf-induced plasma method," have been introduced for commercial processes.

5. Instrumentation.

Again, during the instrumentation phase of the project, it was necessary to strive for simplicity and to limit the number of variables. It was decided to collect data from only two locations. First, a strain gage torsion bridge would be placed on the shaft between the shaft pillow blocks and the spring retainers. Secondly, the gear teeth would be instrumented with strain gages to measure the frictional force during operation.

The torsion bridge was necessary for data from which the normal tooth load could be calculated and finally the Hertzian contact stress determined. It would be a four gage bridge with each gage axis located 45 degrees off the shaft center line. Two gages would be placed at each end of the shaft near the location of the No. 5 arrow on Figure 1 and the gage outputs combined to give a bridge factor of four. Normally this torsion bridge arrangement would be sensitive to bending induced by one motor operation. It would be difficult to arrange the bridge so that this complicated one motor force would not influence the bridge output. This again indicates that two motor operation is preferable. More complete details about a torsion bridge may be found in Perry and Lissner [8].

The measurement of gear tooth forces presents a more difficult problem. A method proposed by Umberger [11] for measuring radial gear tooth forces was adopted and expanded. Since the friction force opposes motion, it acts tangent to the tooth face at the point of contact. This force is not in a pure radial direction. But, by using an engineering approximation, we may consider this friction force as purely radial and be within approximately 6 per cent of the true value. The gears used have a 20 degree pressure angle and the cos 20° is 0.94.

A good theoretical development of the forces on gear teeth including friction was not found. Since the gear tooth is such a complicated structure, normally one of the first simplifying assumptions is to neglect friction. In the case of viscous lubrication this may be allowable since the coefficient of friction is sufficiently small, whereas in solid film lubrication, the friction force may be of the same order of magnitude as the radial force. Assuming a coefficient of friction equal to approximately 0.2 for gold on steel, and a normal tooth load of 200 pounds, we have the following:

$$F(\text{frictional force}) = fN = (0.2)(200) = 40 \text{ lbs.}$$

This may be compared to the radial force of 68 pounds. This extra force definitely affects the stress distribution in the gear tooth and cannot be neglected.

The classical method of gear tooth stress analysis is shown in Figure 12. The frictional force is neglected. The effects of the radial component of force, F_r, are neglected because they are considered small in comparison to the bending forces produced by F_t. The analysis is then continued by transposing the tangential component of force to the tip of a parabolic beam of equal strength, inscribed in the tooth and tangent at the root. This theory is then supplemented by an empirical equation [3], derived by photoelastic methods, to handle stress concentrations at the fillets. This method yields reasonable results when compared to the computer solution described later.

In placing strain gages on the gear teeth, it is well to consider the strain gage matrix. This matrix method was developed by Professor R. E. Newton of the Naval Postgraduate School. By analyzing each proposed method of gage installation with this matrix, it is possible to solve for

that configuration which will produce the highest bridge gage factor for radial forces and introduce the fewest unwanted stray forces. The co-ordinate system used for this strain gage matrix appears in Figure 13.

The three proposed strain gage installations together with the completed matrix solutions are found in Figures 14 and 15. Method No. 1 would appear to measure a radial force (strain) with a bridge factor of four; however, investigation shows that no reading will result. Its complete matrix solution is included to review the method. Using four active gages in the bridge, no strain reading would be produced by radial forces as shown by the matrix solution. A two active gage bridge would react to radial forces, but this would be similar to method No. 3. The radial force producing a moment about the z axis could utilize circuit arrangement B. However, this is not practical because the moment arm or distance between load application and z axis is constantly changing. Also the tangential component of the gear tooth normal force (F_t, page 38 causes moments about the z axis. Strains may reverse signs in gages a and b as the load moves down the tooth face. Thus this arrangement was discarded.

Method No. 2 was an attempt to increase gage output by utilizing the stress concentration factor at the fillets. The gages are located 45 degrees off the tooth center line. Again a bridge factor of four was considered desirable. Its matrix solution showed no bridge output for radial forces. Arguments similar to those proposed for method No. 1 could be developed for this case also.

Method No. 3 shows a bridge output of two, for radial forces, with no addition bridge output due to other than radial forces. Strain gage circuit arrangement A provides this bridge output with gages a and b mounted on either side of the tooth. Gages c and d are idle or compensating gages mounted on other gear teeth about 90 degrees away. By using a

four gage bridge, temperature compensation, although not optimum for arrangement A, is accomplished. By mounting the compensating gages on idle teeth, stray rim stresses are not introduced into the bridge.

The strain gage center line was located at the tooth root. The gage must be low enough to measure loadings at the base of the tooth yet close enough to the load to eliminate, as far as is possible, the effect of strain dispersion in the rim area. This is also the location of a particular stress concentration shown by photoelastic means and the computer program. Pages 42 thru 46 show some of the stress contour plots from the computer program. Some of these plots show this root stress concentration.

Work by Malone [7] had provided a computer program which could be used to check the classical gear tooth stress analysis. Basically, the program uses the finite element method of analysis and works with displacements. A gear tooth shape as indicated in Figure 16 was analyzed by computer. The structure consists of 169 elements. Each element is a triangle or quadrilateral with nodal points at the corners, and centers of the sides. Element sizes were smaller in the area of the tooth and fillets, and gradually increased in size away from the tooth. In this manner, better accuracy was obtained in the areas of interest.

The program output gives nodal point stresses and displacements. Contour plots are constructed and printed. One plot is produced for each, x direction stress, y direction stress, xy shear, maximum normal stress, minimum normal stress, and maximum shear. Some of the contour plots are included in this paper. The ordinate of the plots is distorted during print out. This is done because the area called to be printed is put into a 60 by 100 unit rectangle giving a final plot size of 8½ x 11 inches. The abscissa is automatically made 100 units by the contour plot subroutine.

The ordinate is then adjusted to 60 units. The nodal stresses immediately adjacent to the point of loading (right hand face of tooth) have been eliminated in order to improve the contour plot for the remainder of the tooth and fillet areas. The contour plots closely resemble photoelastic presentations.

The classical computation by the Lewis equations, supplemented by the photoelastic empirical equation [3], yields a maximum stress level of 42,100 psi, tension or compression, in the tooth. This computation is found in the appendix. The various dimensions were scaled off a large diagram of the gear tooth shape for computer analysis. The computer gave a maximum stress of 58,600 psi (compressive) in the left fillet. No impulsive or dynamic loading was assumed. The friction force was neglected for this case and the load assumed acting at the pitch point.

One of the big disadvantages of the classical approach is the fact that friction cannot easily be included. For the case of solid film lubrication, the frictional force may be of the same order of magnitude as the gear separating force, F_r. During motion in one direction, when the frictional force adds to the separating force, the maximum gear tooth stress by computer is 53,400 psi (compressive) as compared to 42,100 psi by the Lewis equations. During motion in the other direction, when the frictional force approximately cancels the separating force, the maximum gear tooth stress by computer is 64,700 psi (compressive) as compared to 42,100 psi by the Lewis equations. As may be seen in the next paragraph, the radial force component has actually helped lower the maximum compressive stress in the left fillet, by producing tensile stresses in this area. The larger the radial force, the lower the maximum compressive in the tooth.

Figure 21 shows the contour plot of minimum normal stress for a load in the radial direction only. This load has the magnitude of the

separating force. It can be readily seen from this plot that the effects of the friction or separating forces are not simple, purely compressive, small, and easily neglected. They are instead, extremely complex, and add significantly to the stress level in the left fillet. In conclusion one could say that classical gear tooth strength theory gives "ball park" values; but in order to fully use gear tooth material, another method of analysis should be used.

The appendix describes the strain gage type and installation. Teflon lead wires were used exclusively because of their outgassing and temperature properties.

6. Testing.

Although the actual testing phase was not started before this writing, a few words along these lines are considered appropriate. The device as constructed is flexible and can be operated in many ways. The method as described below, is just one of many envisioned while working with the device.

In order to retain Hertzian contact stress in the area of the upper design limit, it is necessary to have only one pair of teeth in contact at each test section. Since the gear contact ratio is 1.69, this limits testing to the area of the pitch point. However, one may easily extend this testing area to the entire tooth if the teeth on either side of the testing tooth are removed. In this manner, the full design Hertzian contact load may be applied to the tip or root of the tooth as well as the pitch point.

The lower operating temperature of this device (-250°F) is not attainable with the assembled equipment. Faires [3] lists the nickel requirements for steel in low temperature service. The gears have a maximum of 2 per cent nickel and 2.5 per cent is necessary for safe operation at -75°F. The springs have almost negligible nickel content and therefore are not suited for low temperature service. Preliminary testing at ambient to maximum temperature while at low vacuum is possible until new springs and gears can be procured. Only after procurement of these items may the full design temperature range be tested.

It is suggested that thermocouples be placed on the gear near the test teeth. With these, the temperature in the area of the contact can be monitored.

The stepper motors are limited by their dry film lubricant to about 215,000 cycles. After this, the motors will have to be repaired or replaced. Repair by recoating with new lubricant seems to be the most

practical. Operation during testing then, should be spread out so that all portions of the gearing, etc., receive approximately equal wear.

Because the stepper motors are not good for an infinite number of cycles, small portions of the sputtered gold surface should be tested to failure each time. By not allowing the teeth to mesh through a full contact cycle, the same failure data may be gathered with about one twelfth the motor wear. Additional items will certainly be discovered while working with the equipment.

7. Evaluation.

Once sufficient data have been accumulated for various load levels, a statistical analysis should be performed. Thus for a sputtered film of a certain thickness, one may obtain the probability of the film lasting so long when operated at a certain load level. Eventually, after sufficient data has been analyzed, it may be possible, for a certain film thickness, to relate the Hertzian contact stress to the number of cycles to failure by an empirical equation.

8. Conclusions and Recommendations.

The design, as tested thus far, has proven to be sound. It has been shown that gold may be sputtered upon the test gear shaft. The strain gages have been installed on the gear teeth. With a small amount of additional work, the device will be ready for vacuum chamber testing.

A sputtered gold film possesses all of the important attributes for a good solid film lubricant. It has been shown that the sputtering process is industrially and economically feasible. All that remains is the testing of the film as a lubricant.

The computer program used in the gear tooth strength analysis is a very powerful and important tool. It has shown that some of the assumptions used in classical gear tooth analysis are not absolutely correct. The program listed maximum stresses in gear teeth differing by as much as 36 per cent from the Lewis calculations.

It is recommended that this work be carried to its logical conclusion. The equipment has been almost completely assembled and data should be taken. After testing sputtered gold, silver or other films could be tested. Any number of products are commercially available for testing or new ideas, envisioned by future workers, could be tested.

The computer program has shown that the Lewis equations under estimate the stress by a considerable margin. Additional computer work could be done with the gear tooth shape already existant. New gear tooth sizes could be analyzed. Loads other than at the pitch point and more complicated than only x and y components could be analyzed. New and improved information to predict stress concentration factors could possibly be developed.

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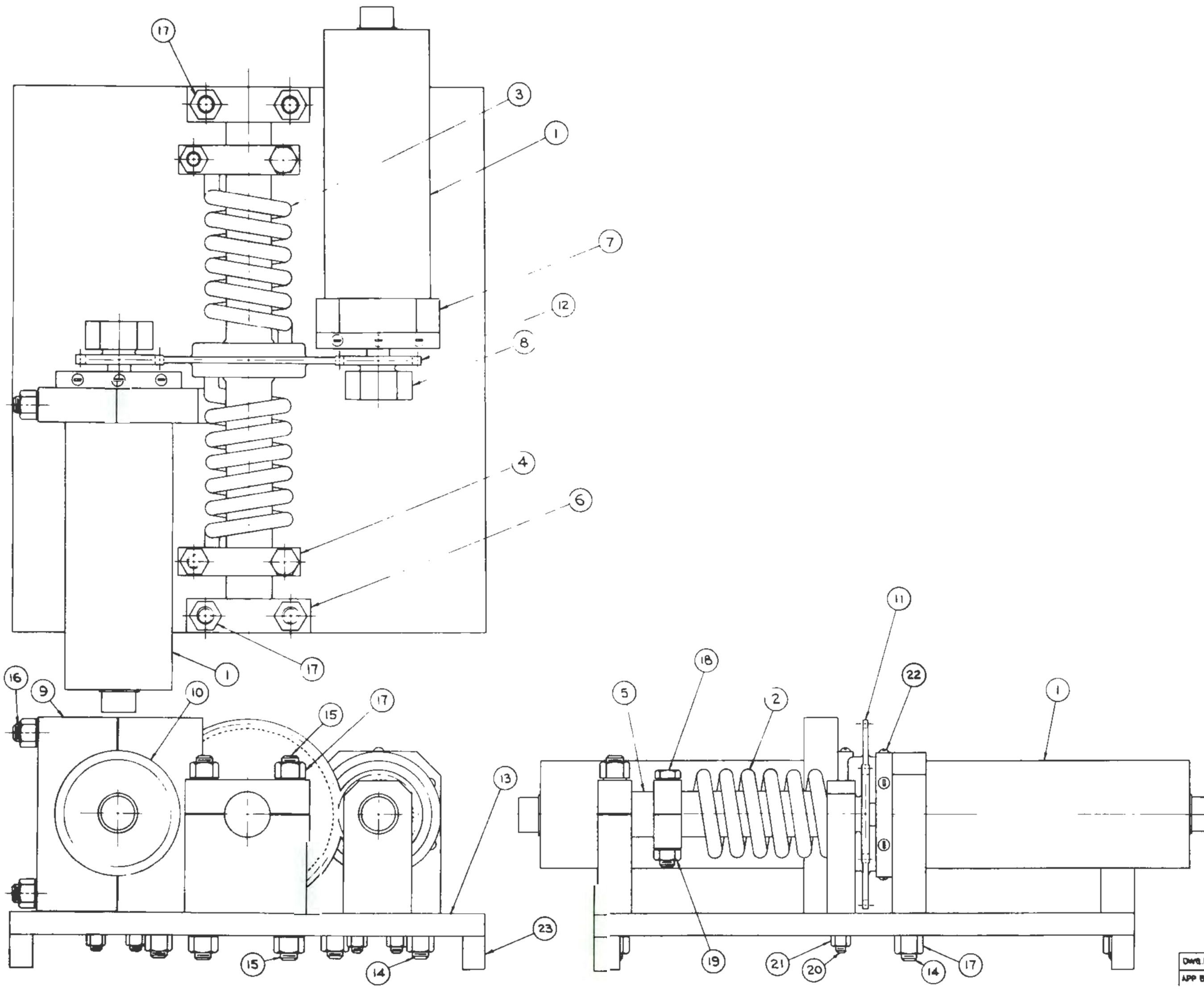

NO.	DISCRIPTION	REQ
23	STAND LEGS	4
22	NO.6-32UNC-2A × 1/2 SST ROUND HD CAP SCR	12
21	1/4-28 UNF-2B SST SEMI-FIN HEX NUT	4
20	1/4-28 UNF-2A × 1 1/2 SST FULL THD STUD	4
19	5/16-24 UNF-2B SST SEMI-FIN HEX NUT	4
18	5/16-24 UNF-2A × 2 SST SEMI-FIN HEX HD BOLT	4
17	3/8-24 UNF-2B SST SEMI-FIN HEX NUT	14
16	3/8-24 UNF-2A × 3 SST FULL THD STUD	2
15	3/8-24 UNF-2A × 2 SST FULL THD STUD	8
14	3/8-24 UNF-2A × 1 1/2 SST FULL THD STUD	4
13	MOUNTING PLATE	1
12	DRIVING PINION	2
11	TEST GEAR	1
10	MOTOR COLLAR	1
9	MOTOR SUPPORT, ADJUSTABLE	1
8	DRIVE GEAR SUPPORT	2
7	MOTOR SUPPORT	1
6	SHAFT PILLOW BLOCKS	2
5	TEST GEAR SHAFT	1
4	SPRING RETAINER	2
3	SPRING - LH	1
2	SPRING - RH	1
1	STEP MOTOR	2

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DRY LUBRICANT TEST STAND ASSEMBLY

DWG. BY [illegible] | APP BY [illegible] | SCALE: 1/2" = 1' | 28 MARCH 1968 | S-504

FIGURE 1. ASSEMBLY DRAWING

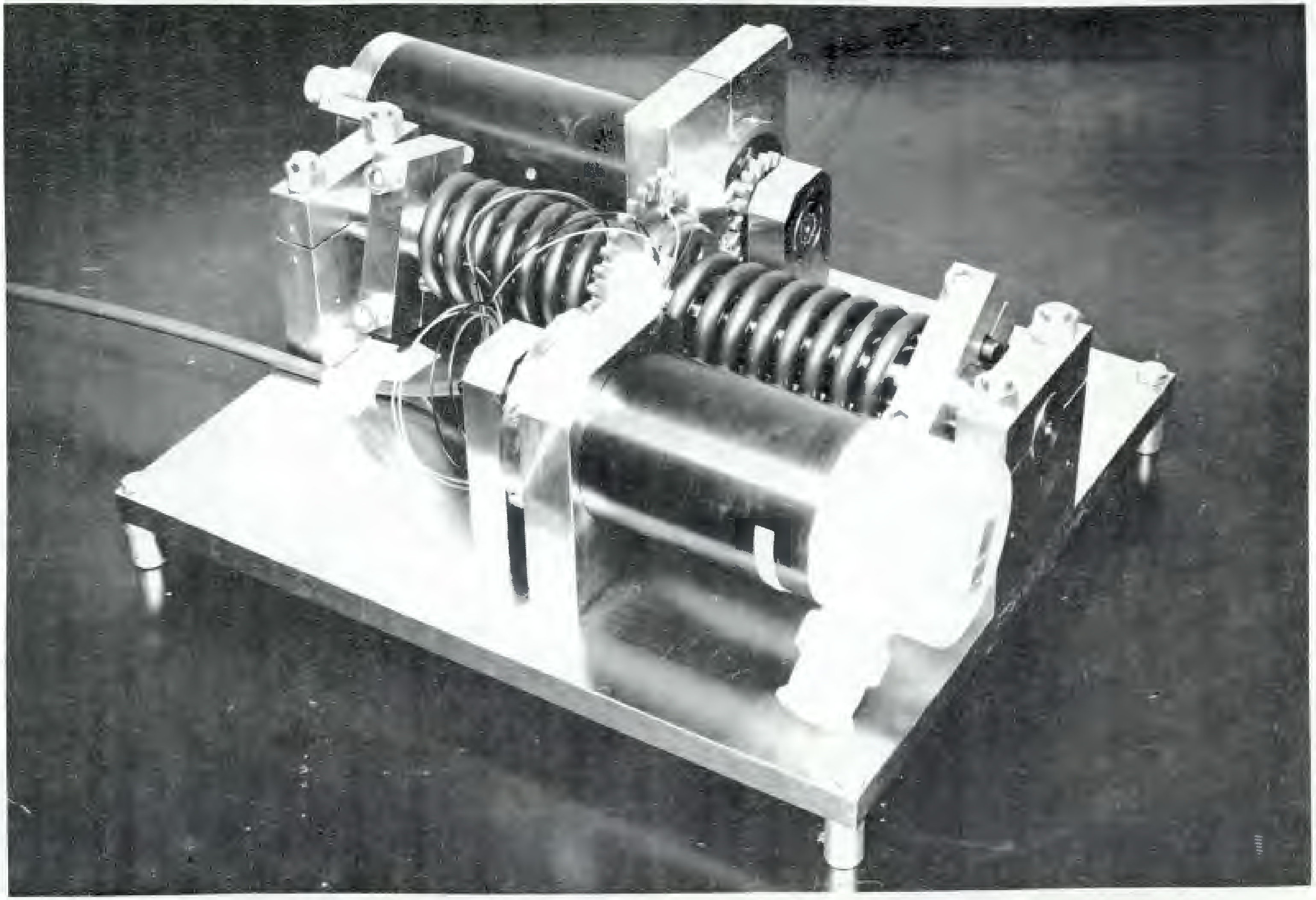
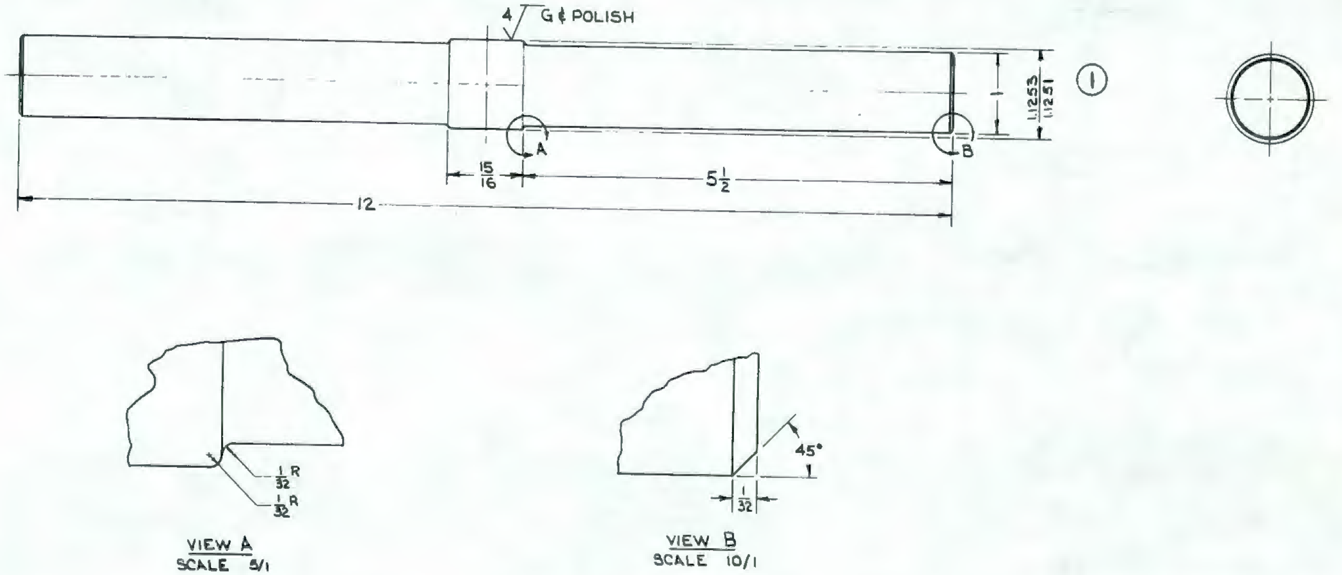


FIGURE 2. PHOTOGRAPH OF ASSEMBLED DEVICE



NO.	CORRECTION	DATE	BY
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MONTEREY CALIFORNIA

TEST GEAR SHAFT

DWG BY MMS
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FIGURE 3. SHAFT DRAWING

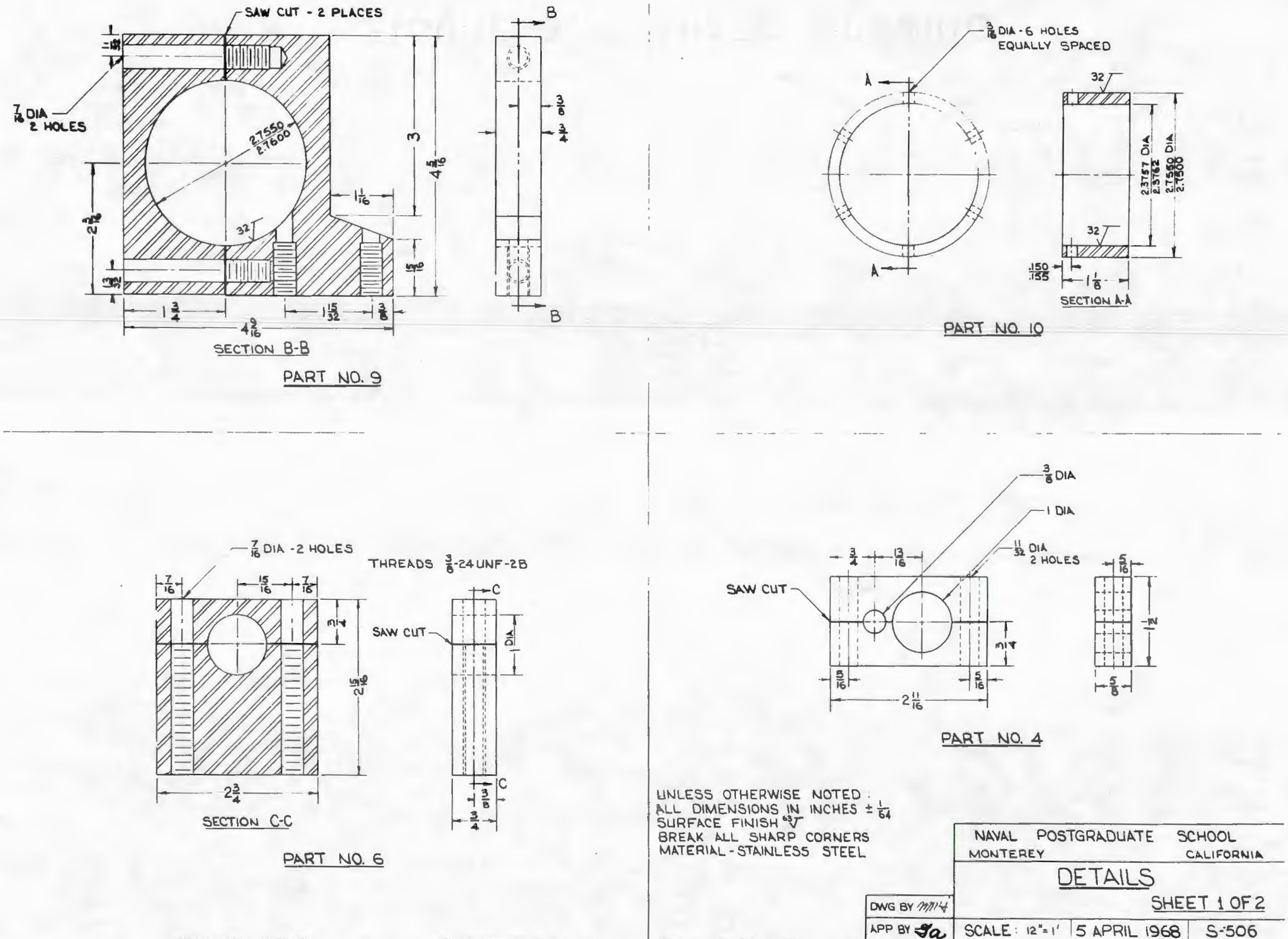


FIGURE 4. DETAILS DRAWING, SHEET 1

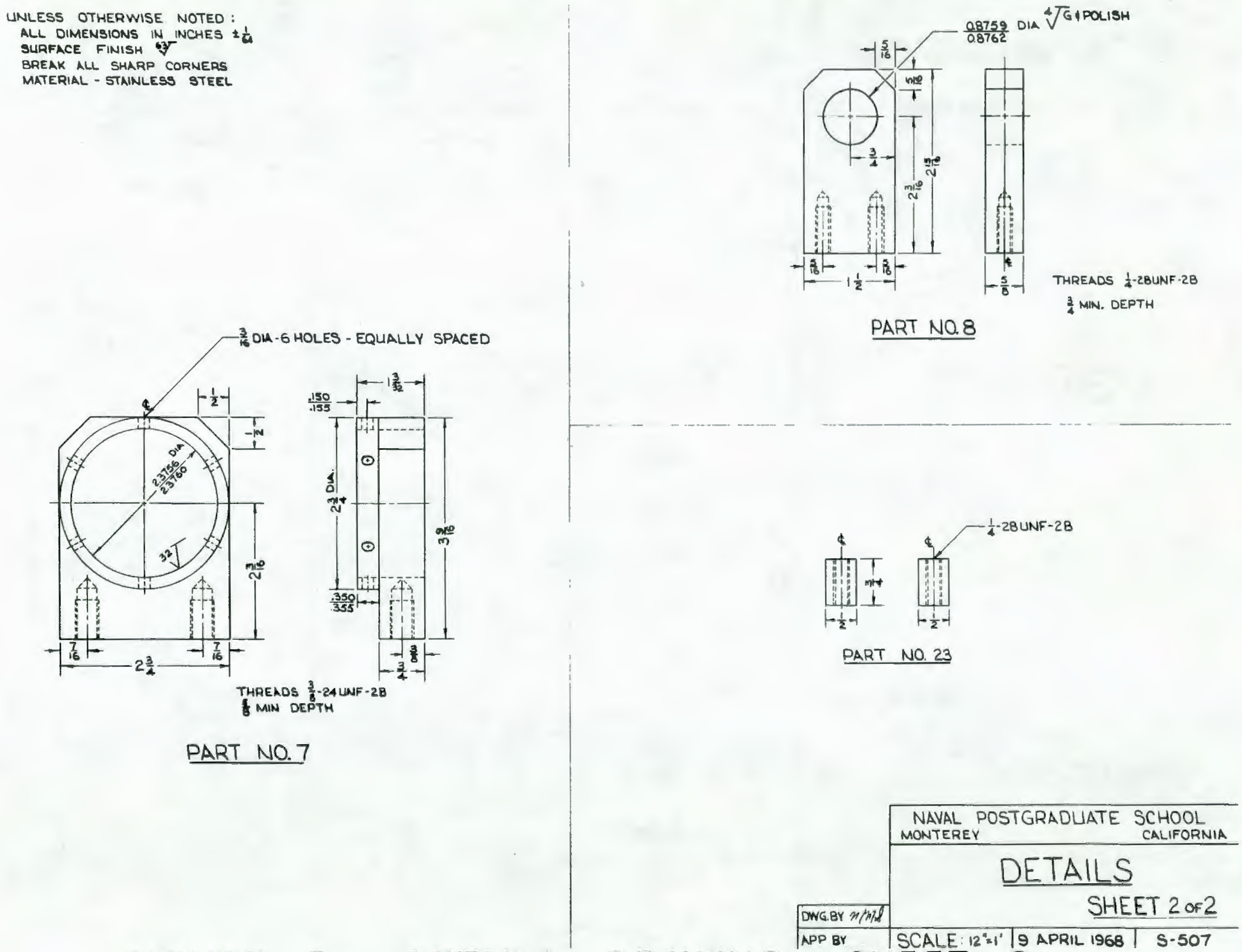


FIGURE 5. DETAILS DRAWING, SHEET 2

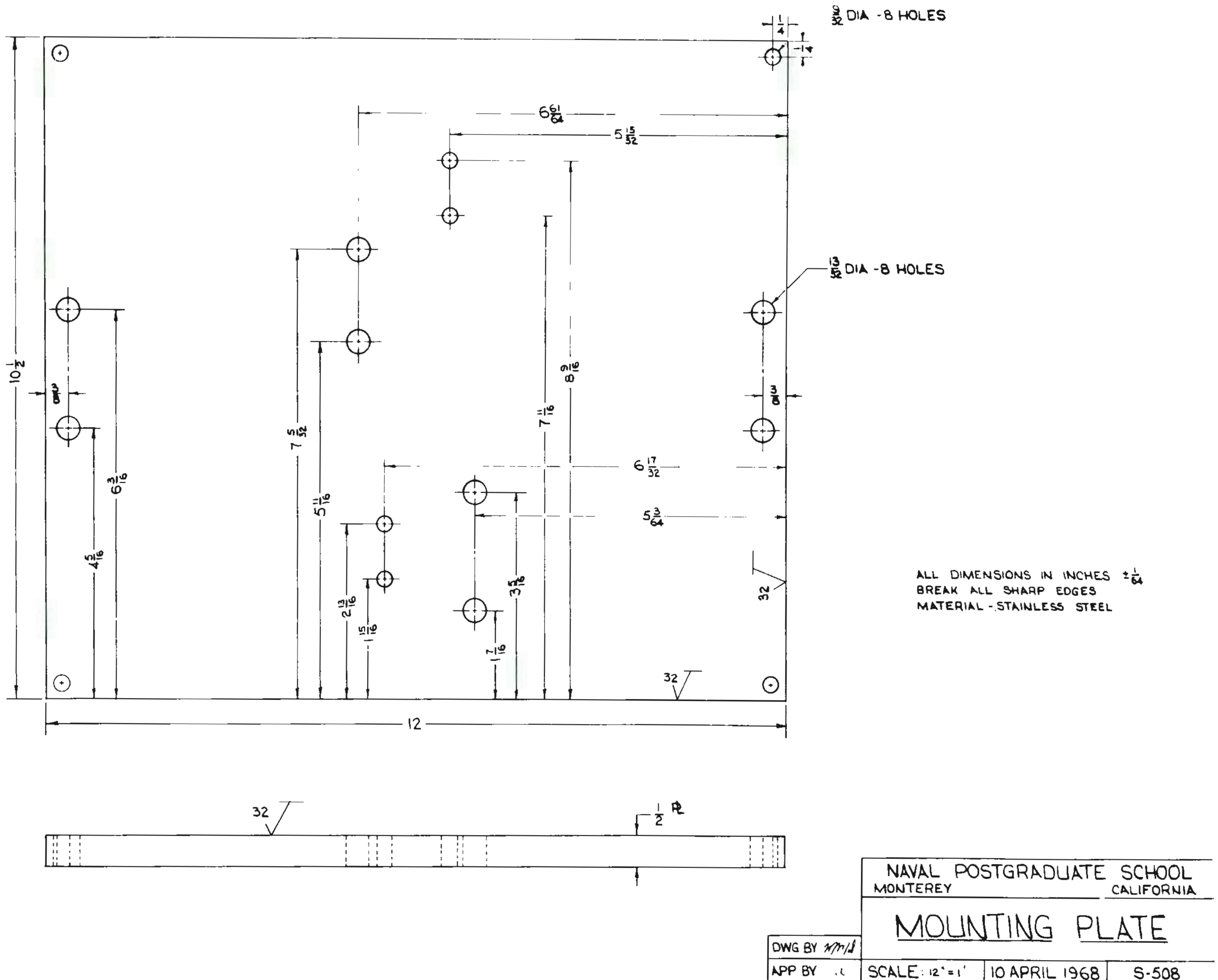


FIGURE 6. MOUNTING PLATE DRAWING

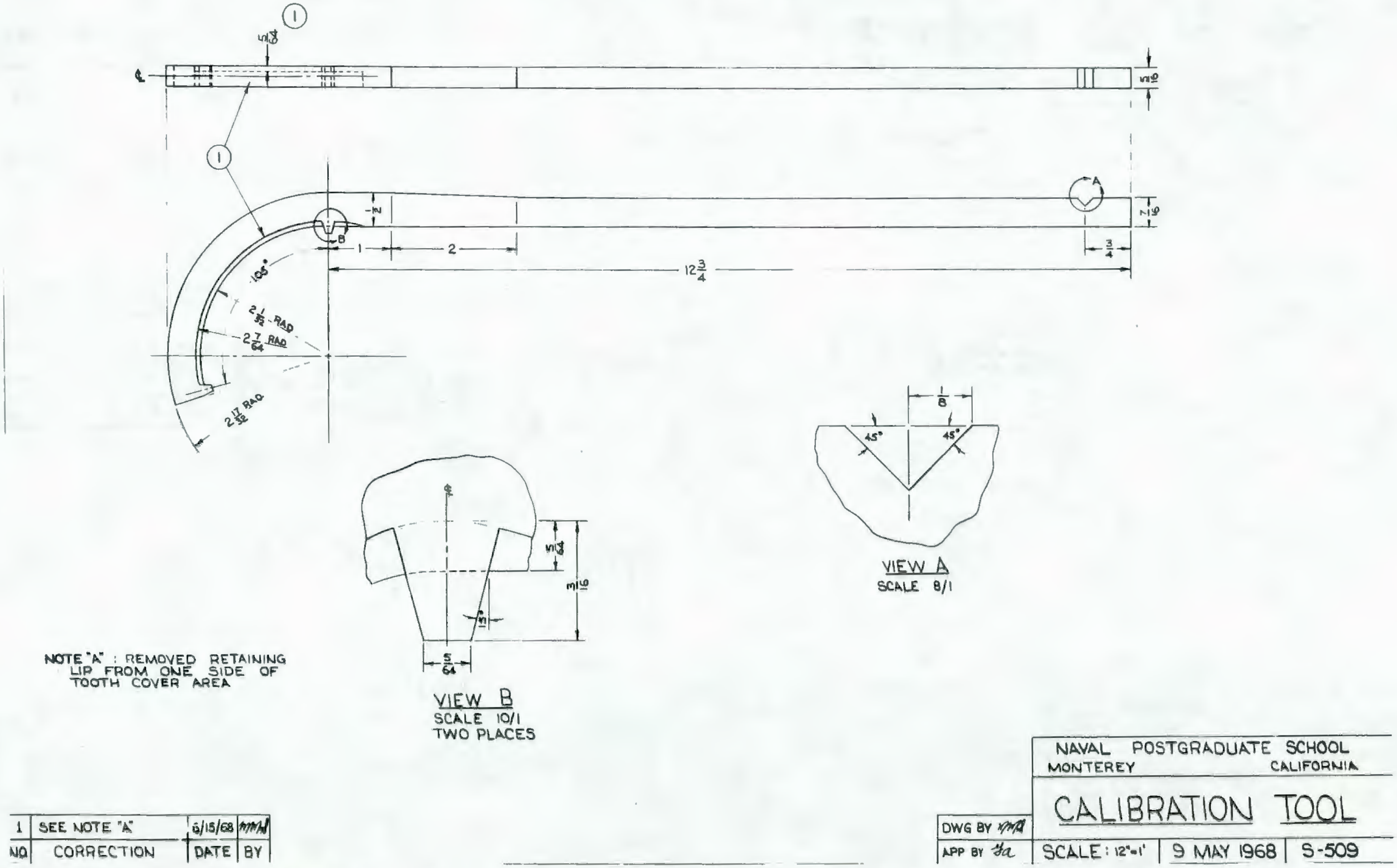
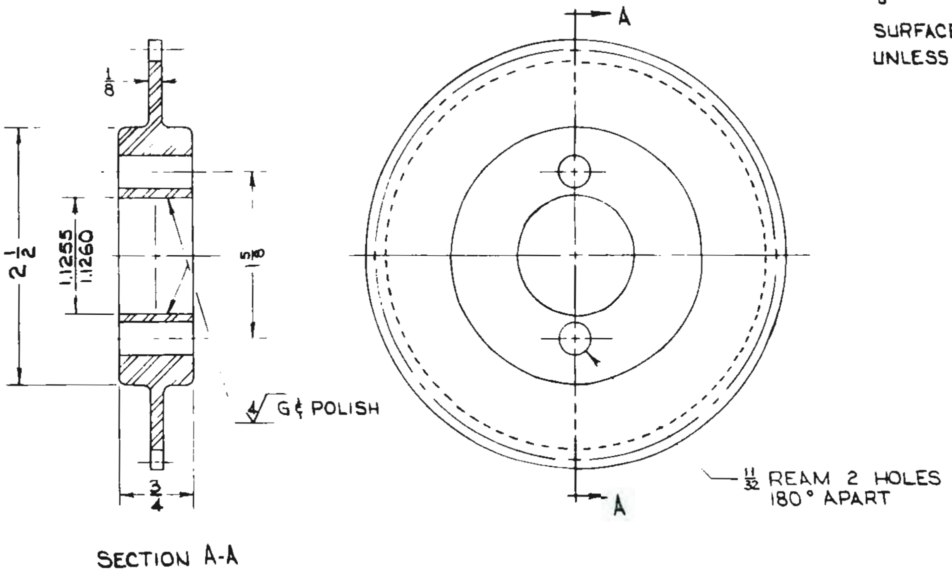


FIGURE 7. CALIBRATION DEVICE DRAWING

ALL DIMENSIONS IN INCHES $\pm \frac{1}{64}$ UNLESS OTHERWISE NOTED
ALL FILLETS AND ROUNDS $\frac{1}{8}$" RAD.

SURFACE FINISH $\overset{63}{\sqrt{}}$
UNLESS OTHERWISE NOTED



GEAR DATA
PITCH DIAMETER = 4"
DIAMETRAL PITCH = 12
PRESSURE ANGLE $\alpha = 20°$
FULL DEPTH INVOLUTE
ADDENDUM (a) = 0.0833"
DEDENDUM (d) = 0.1042"
WHOLE DEPTH = 0.1875"
NO. OF TEETH = 48
MATERIAL - AISI 8640 OQT 1000 OR EQUIVALENT
FACE WIDTH - 0.1250 ± .0005"
MAX. TOOTH ERROR 0.0002"
TOOTH SURFACE FINISH $\overset{20}{\sqrt{}}$

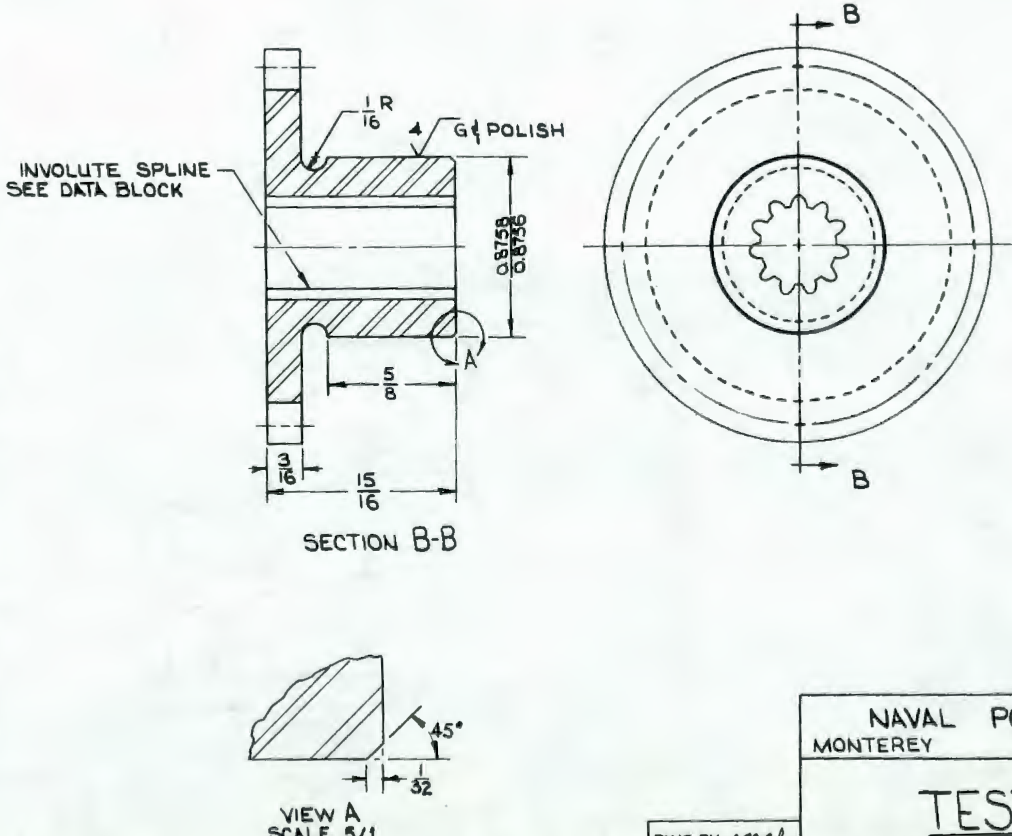
	NAVAL POSTGRADUATE SCHOOL MONTEREY CALIFORNIA		
DWG BY MMH	TEST GEAR		
APP. BY Ga	SCALE : 12" = 1'	6 MARCH 1968	S-502

FIGURE 8. TEST GEAR DRAWING

INVOLUTE SPLINE DATA	
AMERICAN STD. FLAT ROOT SIDE FIT	
PITCH DIAMETER REF	.4583
CIRCULAR PITCH REF	.1309
CIRC. SPACE WIDTH REF	.0684
DIAMETRAL PITCH	24/48
PRESSURE ANGLE	30°
NO. OF TEETH	11
SURFACE FINISH	63√
MAJOR DIA INTERNAL	.5000/.5030
MINOR DIA INTERNAL	.4583/.4633

SPLINES TO MATCH TRANSMISSION SHAFT HUGHES AIRCRAFT DWG. 82577 - 293338

UNLESS OTHERWISE NOTED:
ALL DIMENSIONS IN INCHES $\pm\frac{1}{64}$
SURFACE FINISH 63√



GEAR DATA	
PITCH DIAMETER	1.7500
DIAMETRAL PITCH	12
PRESSURE ANGLE	α = 20°
FULL DEPTH INVOLUTE	
ADDENDUM	0.0833
DEDENDUM	0.1042
WHOLE DEPTH	0.1875
NO. OF TEETH	21
MATERIAL - AISI 8640 OQT @ 1000 OR EQUIVALENT	
FACE WIDTH - 0.1875 ± .0010	
MAX TOOTH ERROR 0.0002	
TOOTH SURFACE FINISH 20√	

NAVAL POSTGRADUATE SCHOOL
MONTEREY CALIFORNIA

TEST PINION

DWG. BY *mml*
APP. BY *Sa*
SCALE: 2" = 1" | 13 MARCH 1968 | S-503

FIGURE 9. TEST PINION DRAWING

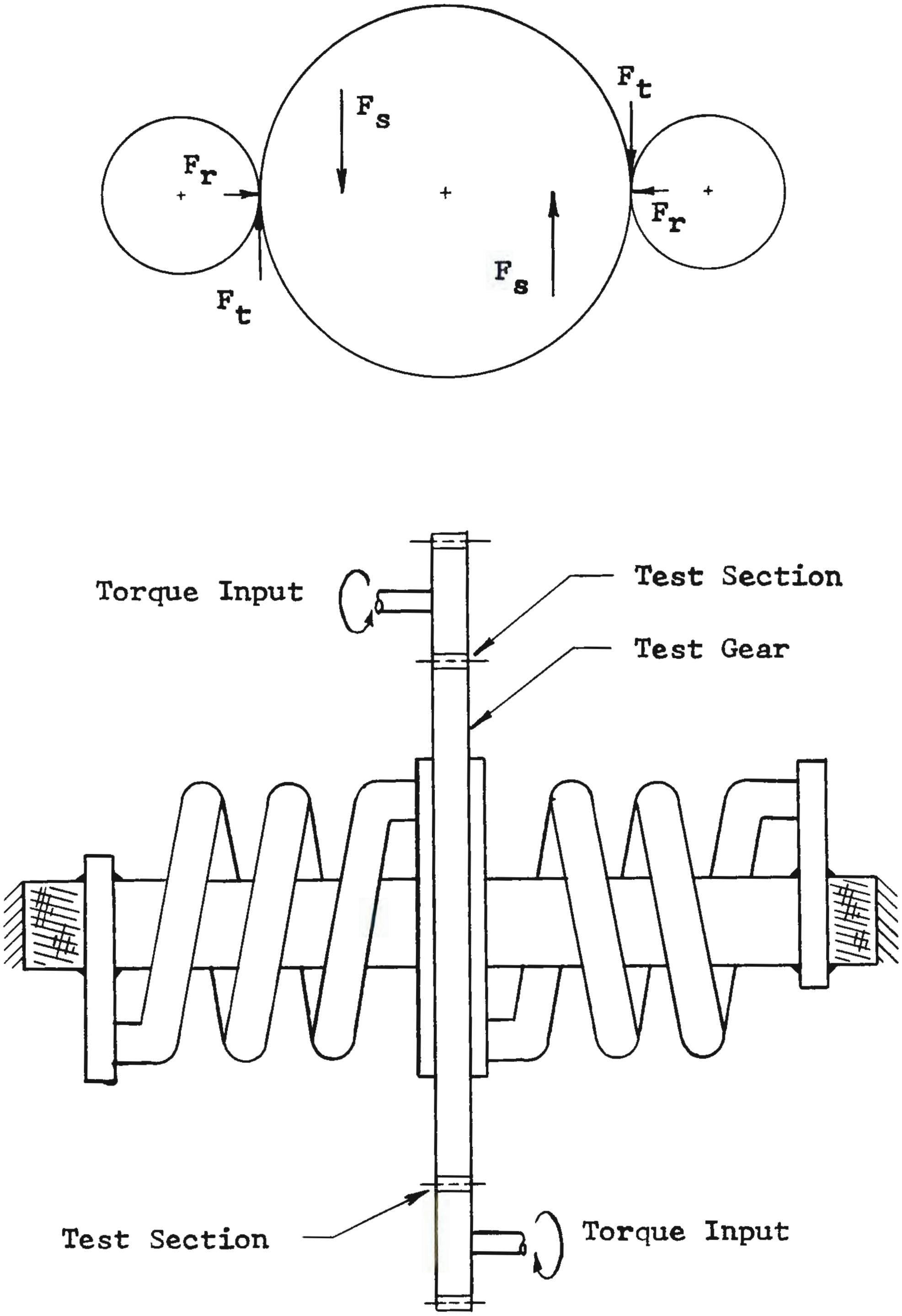


Figure 10. Operating Principles of the Lubricant Tester

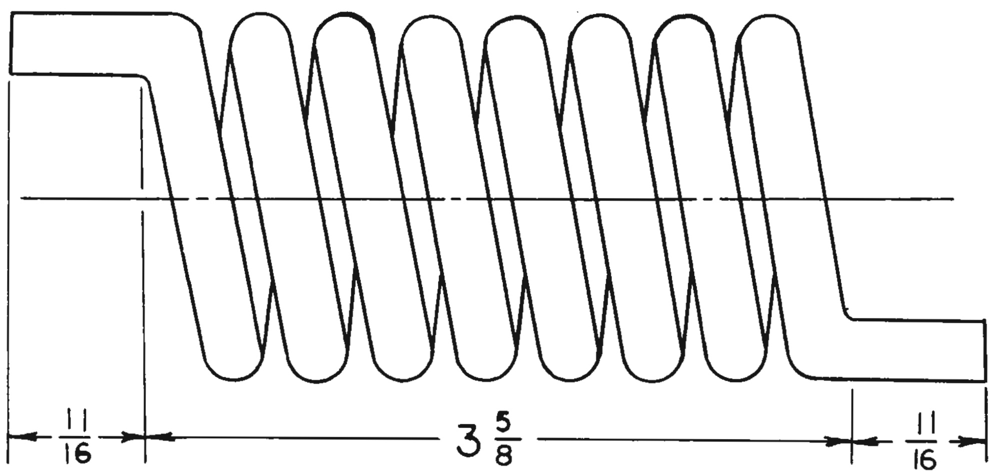


Figure 11. Helical Torsion Spring
Mean Coil Diameter 1-5/8"
Straight Offset Ends
Material S.A.E. 6150
Rod Diameter 5/16"

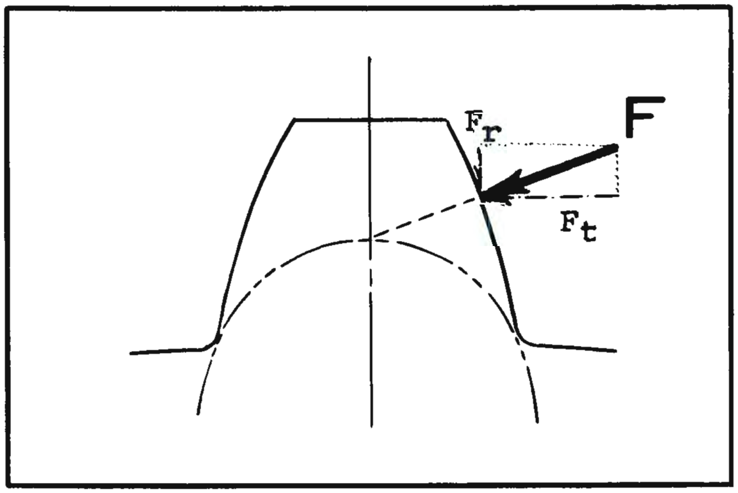


Figure 12. Gear Tooth Forces

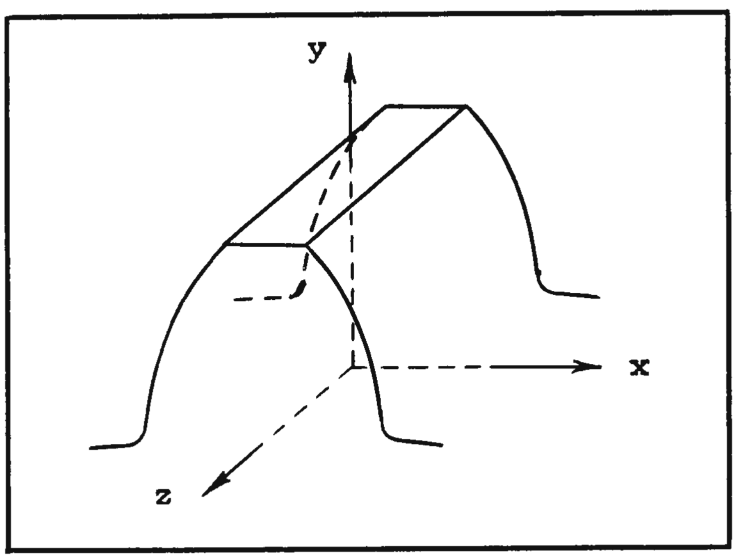
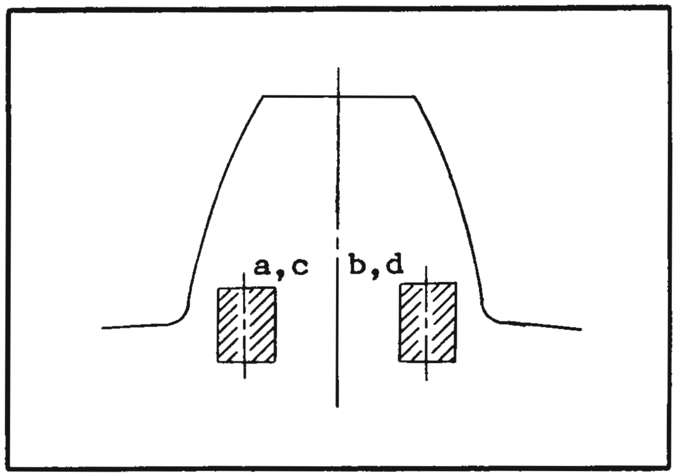
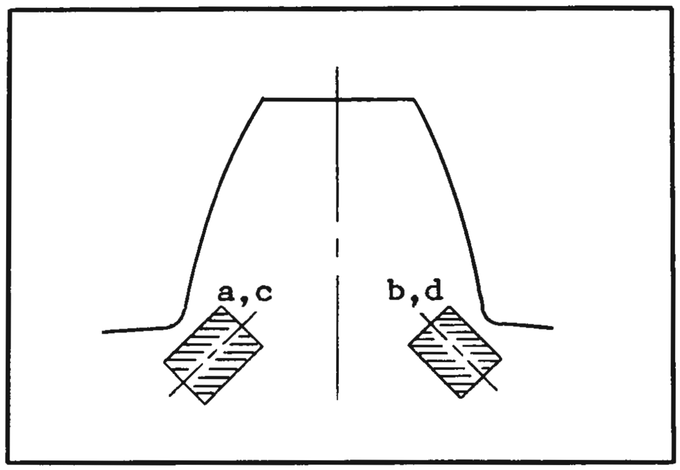


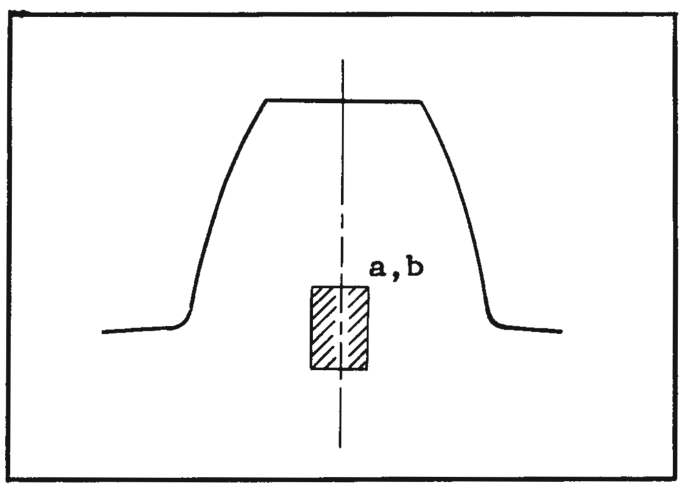
Figure 13. Gear Tooth Coordinate System



Method 1

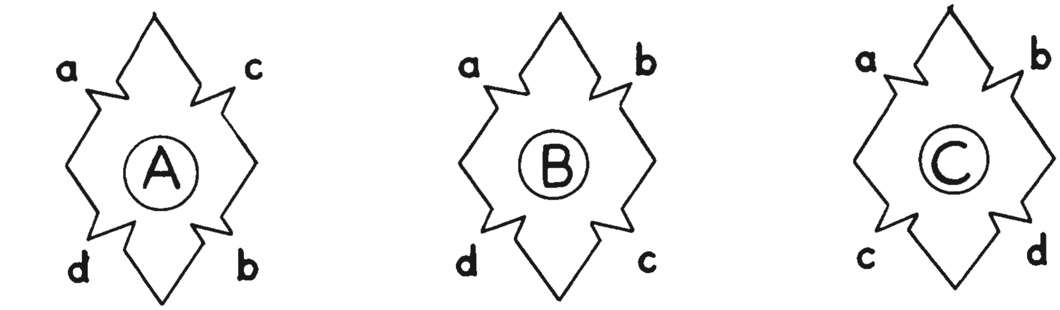


Method 2



Method 3

Figure 14. Proposed Strain Gage Installations



$$
\begin{array}{c}
 \\ F_x \\ F_y \\ F_z \\ M_x \\ M_y \\ M_z
\end{array}
\begin{array}{c}
\begin{array}{cccc} a & b & c & d \end{array} \\
\begin{bmatrix}
0 & 0 & 0 & 0 \\
1 & 1 & 1 & 1 \\
0 & 0 & 0 & 0 \\
-1 & -1 & 1 & 1 \\
0 & 0 & 0 & 0 \\
-1 & 1 & -1 & 1
\end{bmatrix}
\end{array}
\begin{array}{c}
\begin{array}{ccc} A & B & C \end{array} \\
\begin{bmatrix}
-1 & -1 & -1 \\
-1 & 1 & 1 \\
1 & -1 & 1 \\
1 & 1 & -1
\end{bmatrix}
\end{array}
\begin{array}{c}
 \\ a \\ b \\ c \\ d
\end{array}
=
\begin{array}{c}
\begin{array}{ccc} A & B & C \end{array} \\
\begin{bmatrix}
0 & 0 & 0 \\
0 & 0 & 0 \\
0 & 0 & 0 \\
4 & 0 & 0 \\
0 & 0 & 0 \\
0 & 4 & 0
\end{bmatrix}
\end{array}
\begin{array}{c}
 \\ F_x \\ F_y \\ F_z \\ M_x \\ M_y \\ M_z
\end{array}
$$

RESULTS NO.1

$$
\begin{array}{c}
\begin{array}{ccc} A & B & C \end{array} \\
\begin{bmatrix}
0 & -4 & 0 \\
0 & 0 & 0 \\
0 & 0 & 0 \\
2(1-\mu) & 0 & 0 \\
0 & 0 & -4 \\
0 & 2(1-\mu) & 0
\end{bmatrix}
\end{array}
\quad
\begin{array}{c}
 \\ F_x \\ F_y \\ F_z \\ M_x \\ M_y \\ M_z
\end{array}
\quad
\begin{array}{c}
\begin{array}{ccc} A & B & C \end{array} \\
\begin{bmatrix}
0 & 0 & 0 \\
-2 & 0 & 0 \\
0 & 0 & 0 \\
0 & 2 & 2 \\
0 & 0 & 0 \\
0 & 0 & 0
\end{bmatrix}
\end{array}
$$

RESULTS NO.2 RESULTS NO.3

FIGURE 15. STRAIN GAGE BRIDGE CIRCUITS AND MATRIX SOLUTIONS

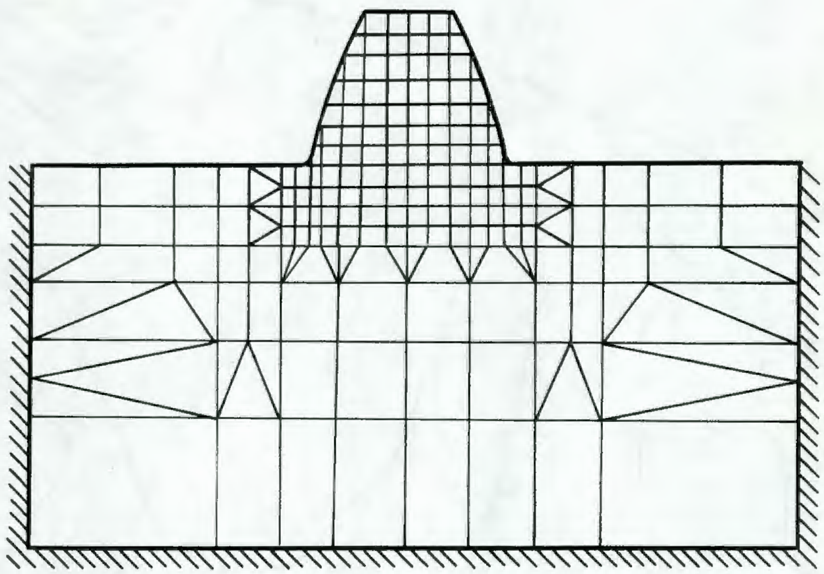


Figure 16. Gear Tooth Shape For Computer Analysis

Sigma Maximum.
Normal Load.
No Friction.
To Determine Stress
Multiply Contour Plot
Number By 4 ksi.

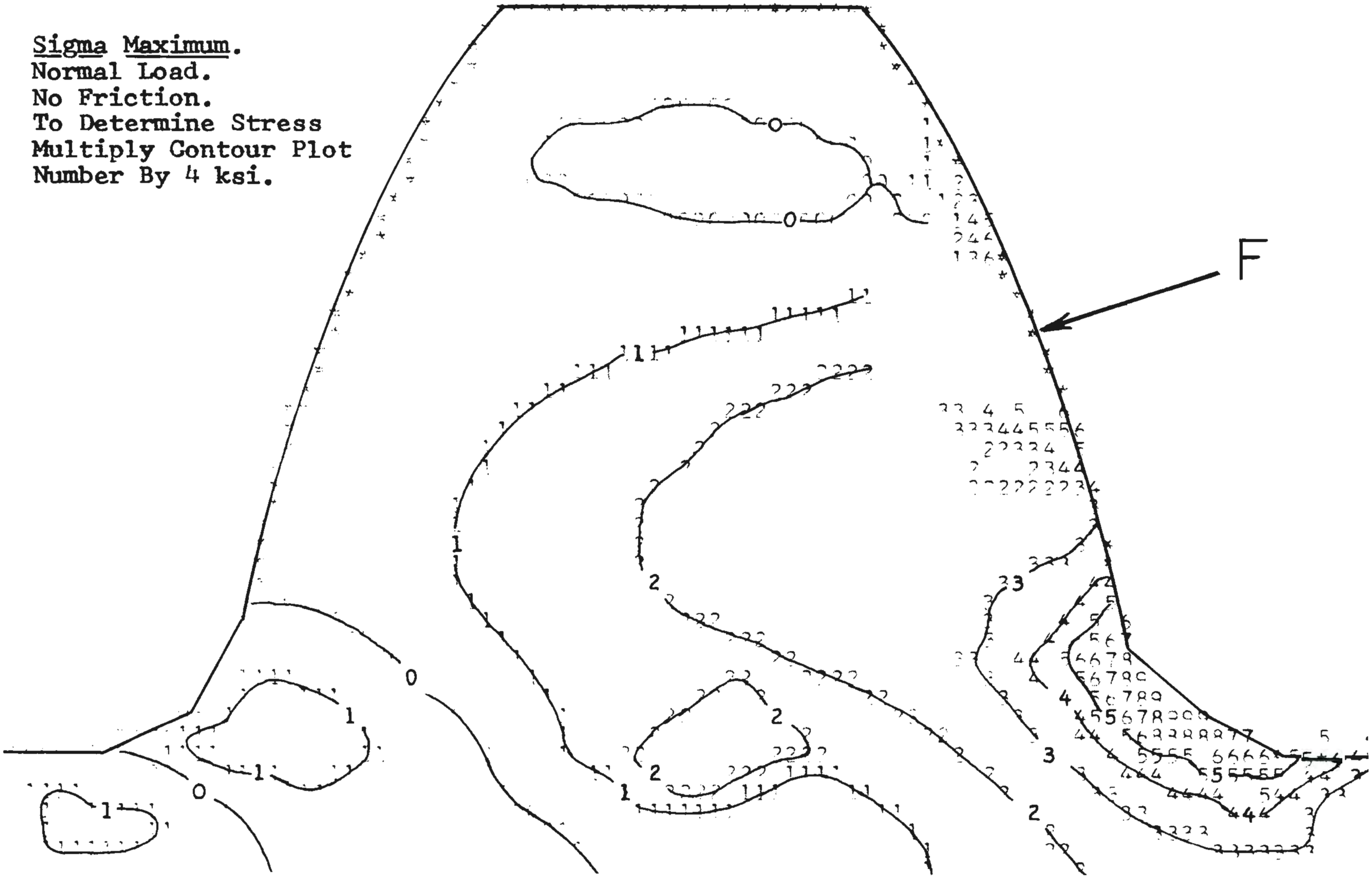


FIGURE 17. MAXIMUM NORMAL STRESS - NORMAL LOAD

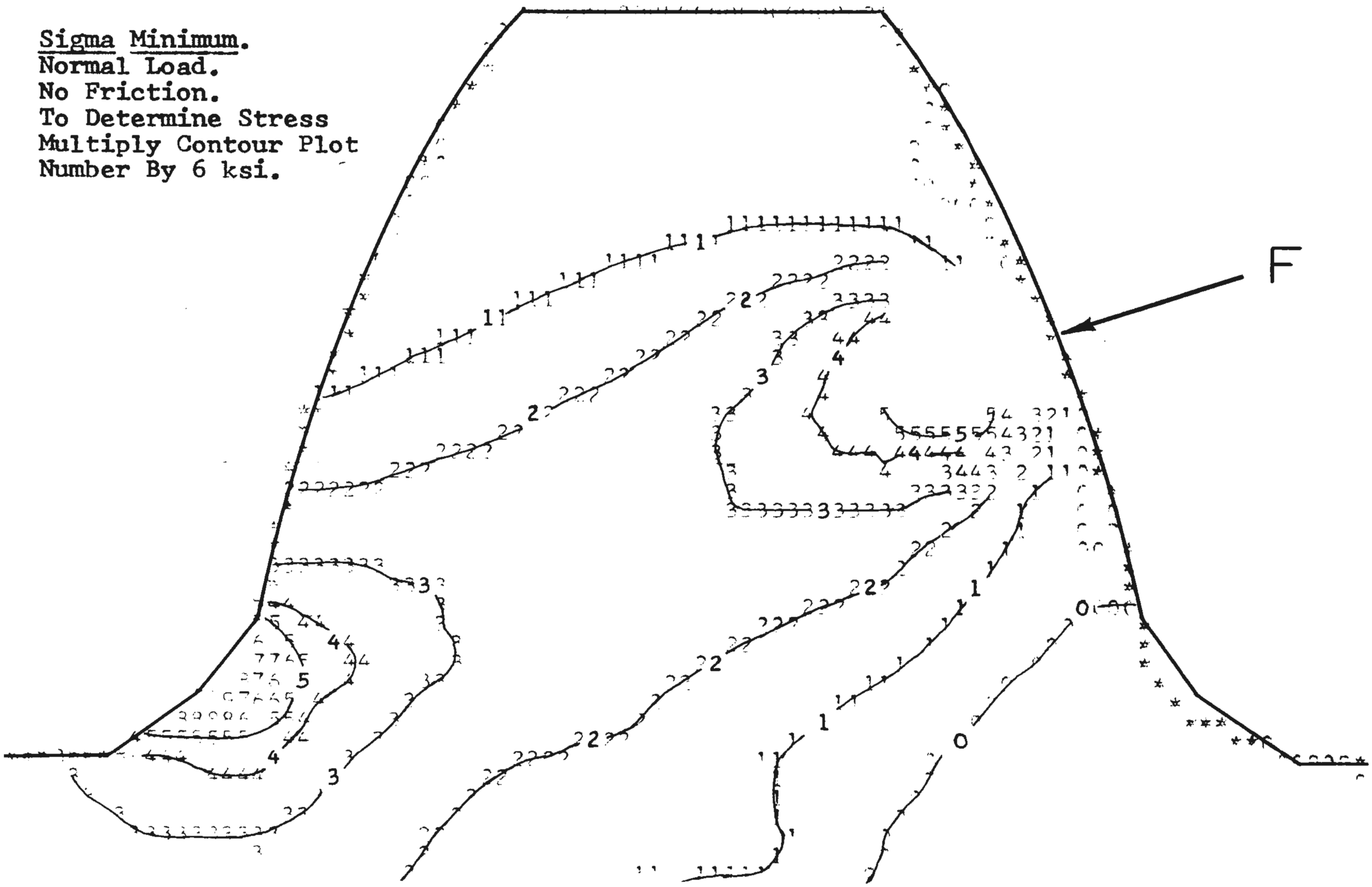


FIGURE 18. MINIMUM NORMAL STRESS - NORMAL LOAD

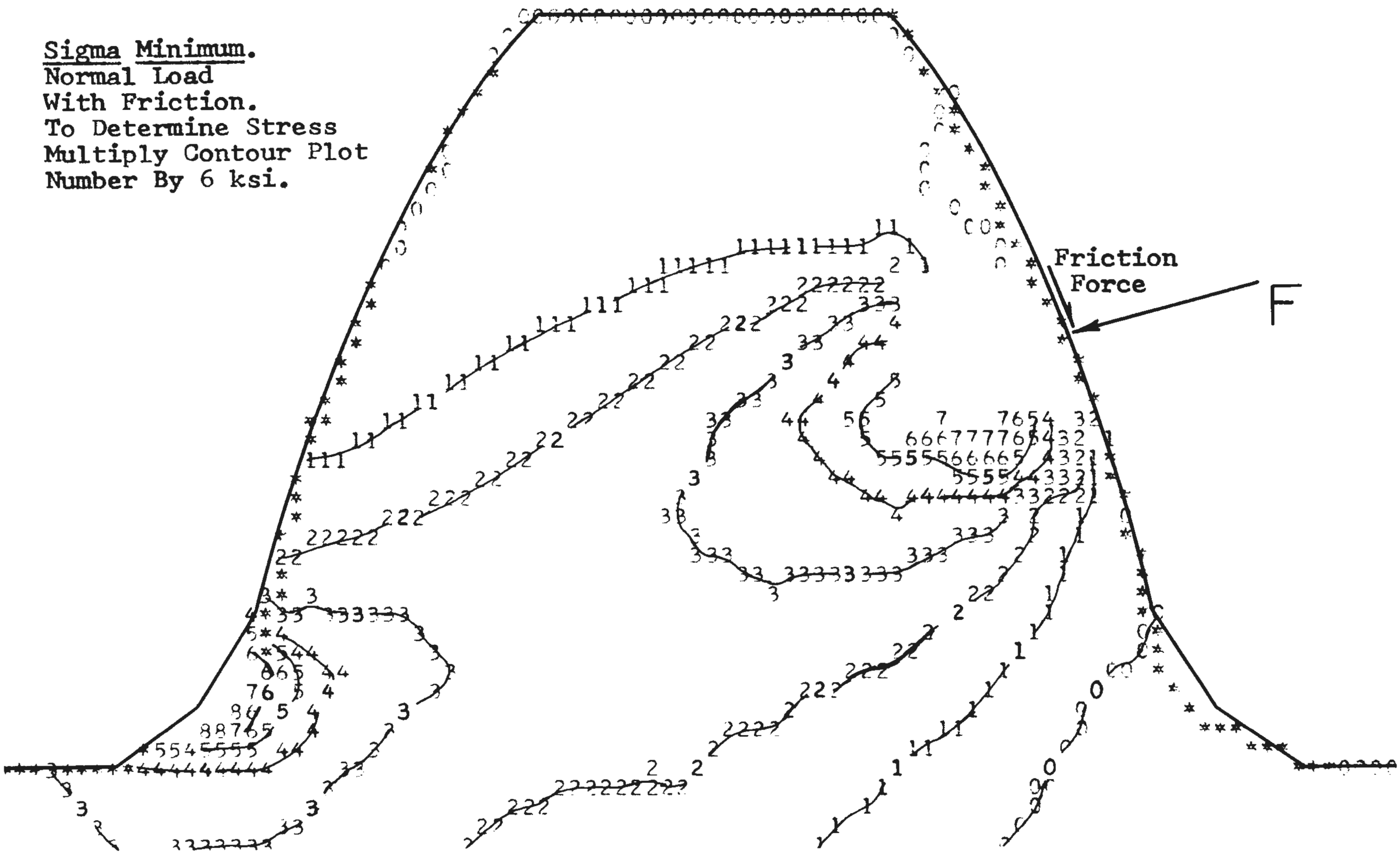


FIGURE 19. MINIMUM NORMAL STRESS - NORMAL LOAD, FRICTION

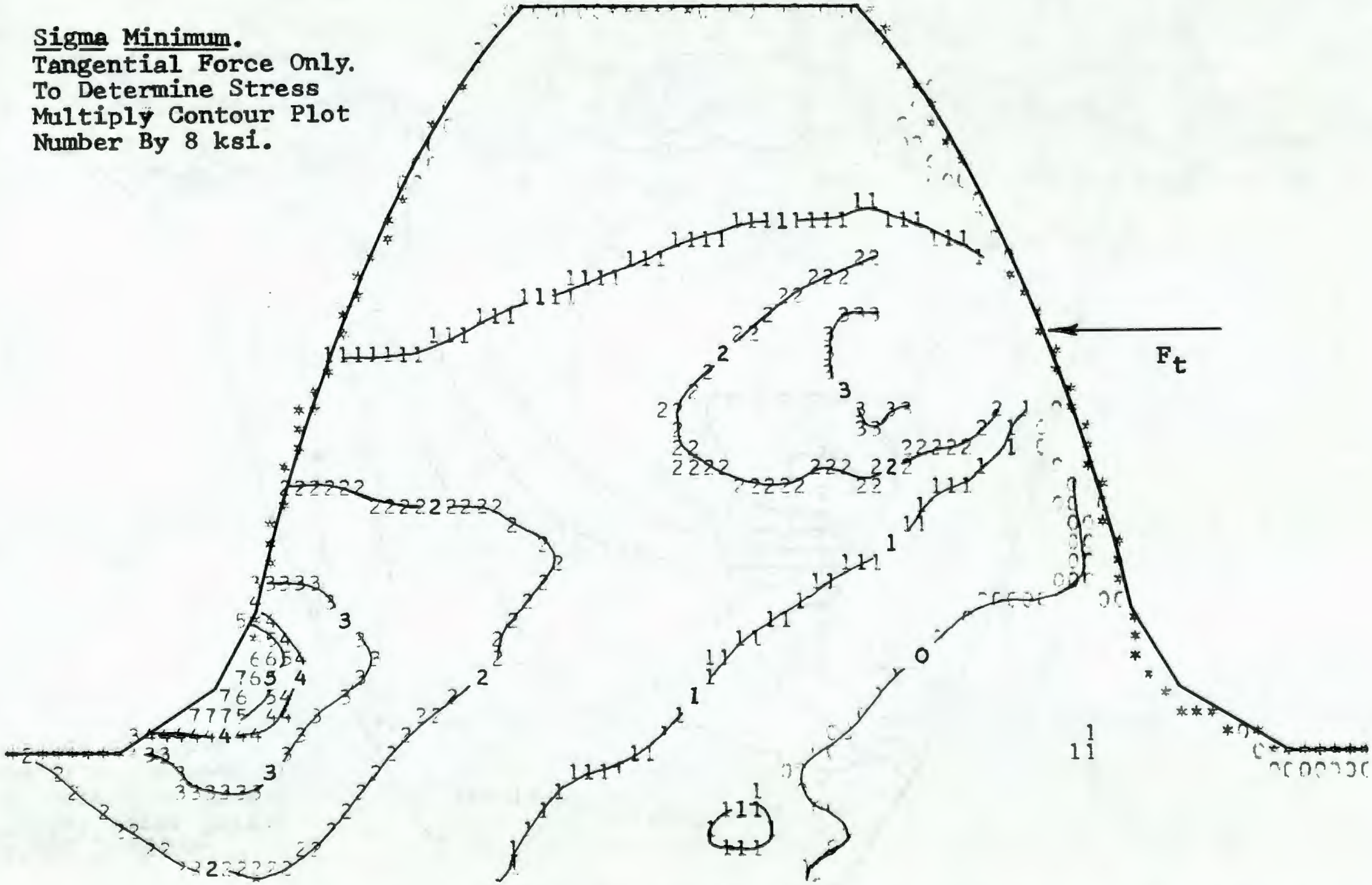


FIGURE 20. MINIMUM NORMAL STRESS - TANGENTIAL FORCE ONLY

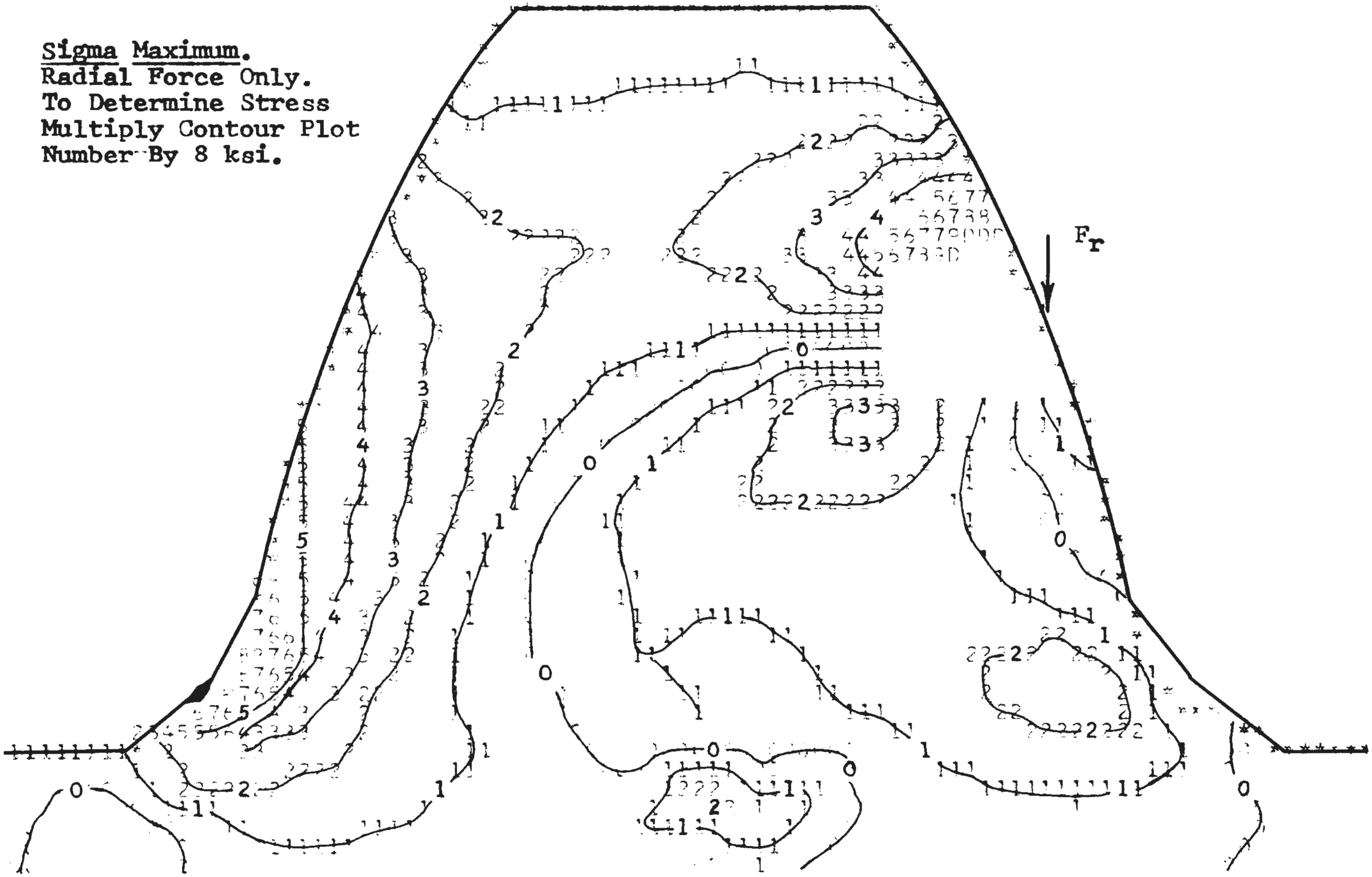


FIGURE 21. MAXIMUM NORMAL STRESS - RADIAL FORCE

APPENDIX

1) Gear Tooth Strain Gage Type and Installation

2) Gear Tooth Calculations Using the Lewis Equations

Gear Tooth Strain Gages

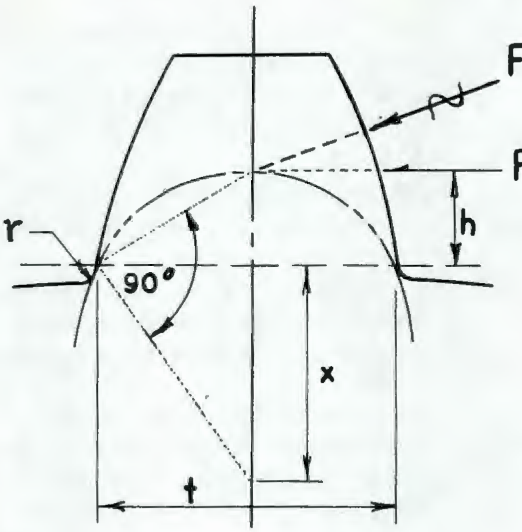
The strain gages and bonding cement were manufactured by Baldwin-Lima-Hamilton Electronics, Inc. The gages are type FAE-03N-12S6 with a gage factor of 1.94 $\pm$ 2%. They have a resistance of 120.0 $\pm$ .2 ohms and were produced as part of Lot # 252. The gages were installed with EPY-400 bonding cement and cured at 300°F for three hours. This treatment should render the strain gage bridge effective between -250°F and +300°F.

GEAR TOOTH STRESS BY LEWIS EQUATIONS

$$Y = \frac{2 x P_d}{3}$$

$$K = 0.18 + \left(\frac{t}{r}\right)^{0.2}\left(\frac{t}{h}\right)^{0.4}$$

$$F_t = \frac{sbY}{KP_d}$$



WHERE

Y = LEWIS FORM FACTOR

P_d = DIAMETRAL PITCH

K = STRESS CONCENTRATION FACTOR FOR FILLETS

F_t = TANGENTIAL COMPONENT OF NORMAL LOAD

b = TOOTH FACE WIDTH

s = MAXIMUM TOOTH STRESS

$$Y = \frac{(2)(0.13)(12)}{3} = 1.04$$

$$K = 0.18 + \left(\frac{.162}{.027}\right)^{.2}\left(\frac{.162}{.051}\right)^{.4} = 2.45$$

$$S = \frac{F_t K P_d}{bY} = \frac{(186)(2.45)(12)}{(.125)(1.04)} = 42,100 \text{ psi}$$

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13. ABSTRACT

The design, construction, and instrumentation of a solid film lubricant tester are discussed. The testing phase and evaluation of test data are mentioned. The device was constructed to test sputtered gold as a gear tooth lubricant for low vacuum operation. The unit is small and self-sustained for operation entirely within the vacuum chamber with only external electrical leads. Methods of measuring gear tooth frictional forces are discussed. A computer program is used to check the results of classical gear tooth strength theory.

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